Assessment Cover

Module No:	**ENGR7015**	Module title:	**Advanced Vehicle Dynamics**

Assessment title :	**Portfolio Coursework**

Due date and time**:**	13:00hrs Friday 23rd December 2022.

Estimated total time to be spent on assignment:	**80 hours per student**

LEARNING OUTCOMES

On successful completion of this module, students will be able to achieve the module following learning outcomes (LOs): *LO numbers and text copied and pasted from the module descriptor*.
LO 1: Demonstrate a systematic and ordered understanding of vehicle performance backed up with analytical determination of performance;
LO 2: Perform complex analyses on whole vehicle performance and critically assess one vehicle against another.
LO 3: Critically examine current methods of vehicle performance evaluation
LO 4: Use numerical and analytical techniques to critically analyse different aspects of vehicle performance.
LO 5: Make rational and informed decisions to select methods for improving vehicle performance over a wide range of situations.
LO 7: Undertake independent learning of ‘new vehicle dynamics related problems’ at an advanced level

Engineering Council AHEP4 LOs assessed (from S1 2022-23):	
M1	Apply a comprehensive knowledge of mathematics, statistics, natural science and engineering principles to the solution of complex problems. Much of the knowledge will be at the forefront of the particular subject of study and informed by a critical awareness of new developments and the wider context of engineering
M2	Formulate and analyse complex problems to reach substantiated conclusions. This will involve evaluating available data using first principles of mathematics, statistics, natural science and engineering principles, and using engineering judgment to work

	with information that may be uncertain or incomplete, discussing the limitations of the techniques employed
M3	Select and apply appropriate computational and analytical techniques to model complex problems, discussing the limitations of the techniques employed
M4	Select and critically evaluate technical literature and other sources of information to solve complex problems
M17	Communicate effectively on complex engineering matters with technical and non-technical audiences, evaluating the effectiveness of the methods used

STUDENT NAME(S)

Student No:	Student Name:	Group Name and Number:
1.	**PLEASE DO NOT ADD YOUR NAME AS SUBMISSION AND ASSESSMENT IS ANONYMOUS**	

Statement of Compliance *(please tick to sign)*

I declare that the work submitted is my own and that the work I submit is fully in accordance with the University regulations regarding assessments (*www.brookes.ac.uk/uniregulations/current*)

Table of Contents

Section 1- Tire Model 4
- **Introduction** 4
- **F-Tire Model** (Flexible Ring Tyre Model) 4
- **RMOD-K Model** 5
- **Tame tyre** 6
- **Comparison Sheet** 8

Section – 2 Suspension Optimisation 9
- **Introduction** 9
- **Suspension optimisation using SDOF** 9
- **Suspension optimisation using 2DOF** 10
- **Suspension optimisation using 4DOF** 13
- **Conclusion** 14

Section - 3 Derivatives and Bicycle Model 14
- **Introduction** 14
- **Derivative Analysis** 15
- **Derived Parameters** 15
- **ADAMS Model** 16
- **Result** 17
- **Comparison between Adams model and Derivative Analysis** 17
- **Conclusion** 18

References 18

Section 1- Tire Model

Introduction

Vehicle dynamics is the study of the behaviour of a vehicle in response to driver inputs, road conditions and other factors. Tyres are a crucial component of vehicle and are the only point of contact between the road surface and the vehicle chassis. Tyres are responsible for most of the forces acting on the vehicle, therefore a vehicle dynamist needs to understand its behaviour. Tyre models are mathematical models which are used to describe the behaviour of tyres and their interaction with road surfaces. Tyre models provide information about the effect of forces on the vehicle for different road surfaces. Using this information vehicle dynamists can optimize the design for better handling and performance.

F-Tire Model (Flexible Ring Tyre Model)

The F-Tire is an advanced model for Tyre force elements as it considers modules for tread wear, thread geometry, thermal modules etc. to accurately depict the road contact, which is used for the study of vehicle ride comfort and durability. A major goal for the development of F-Tire was to be easy to implant, have nonlinear attributes with a higher frequency range of up to 120Hz, be able to observe road translation inclination, lower computational power etc.

The model was developed as a 3D vibrational model made of flexible elements attached with springs in a particular pattern where the belt elements considered are elastically suspended on the rim with the stiffness and damping elements distributed in radial, tangential, and lateral directions. The rings are approximated by a finite number of point masses where the belt elements are attached to nearby elements. During the pre-processing step, all stiffness and damping factors are calculated.

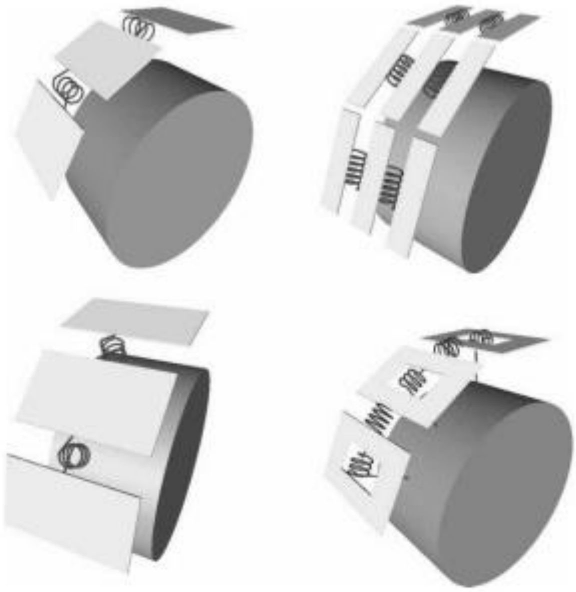

Figure 1 F-tire representation (Lunger, et al., 2007)

For each belt element, some tread ribs are massless and have non-linear stiffness and damping properties in all three directions. The road profile, locus and belt element orientation are the parameters which affect the radial deflections. The tangential and lateral deflections can be determined by the sliding velocity and sliding coefficient. The sliding velocity is the point velocity projected on the road profile in the tangent plane.

In this model, important components such as forces and moments operating, are estimated by integrating the forces on the belt's elastic foundation which makes this model accurate. Various input quantities are specified, including rim positions, the angular orientation of the rim, translational rim velocity, angular rim velocity, road profile, skid number, rolling circumference, rim diameter, and tread width.

The F-Tire is mainly focused towards the ride comfort simulation and Road load predictions for irregular road profiles. Since this model is non-linear and dynamic, it allows detailed studies of vehicle dynamics for various road profiles considering the suspension control system design.

This Tyre model is available in most multi-body simulation packages.

RMOD-K Model

RMOD-K model system is used in simulations for driving dynamics, driving comfort and durability tests with virtual vehicle models that can be carried out at the early stage of vehicle development. The RMOD-K was developed to determine the Steady-state tyre response for different excitation frequencies and different driving inputs.

The model features four main blocks in its development: The Driving dynamics model, structure dynamics model, road surface models and tyre characteristics.

The driving dynamics model allows the representation of the stationary performance characteristic of the tyre. Whereas the structure dynamics model helps with the calculation of the tyre behaviour. The road surface model is required to integrate the road sections and describe the actual tracks. The tyre characteristic model includes linearized model types using which the modal system for a loaded or unloaded tyre can be determined.

RMOD-K model features a flexible belt connected to the rim with a sidewall model with pressurised air. Road contact is made achievable by a supplementary sensor layer added to a belt which can be single or multi-layered that interact with one another. The normal and frictional forces are calculated at sensor locations.

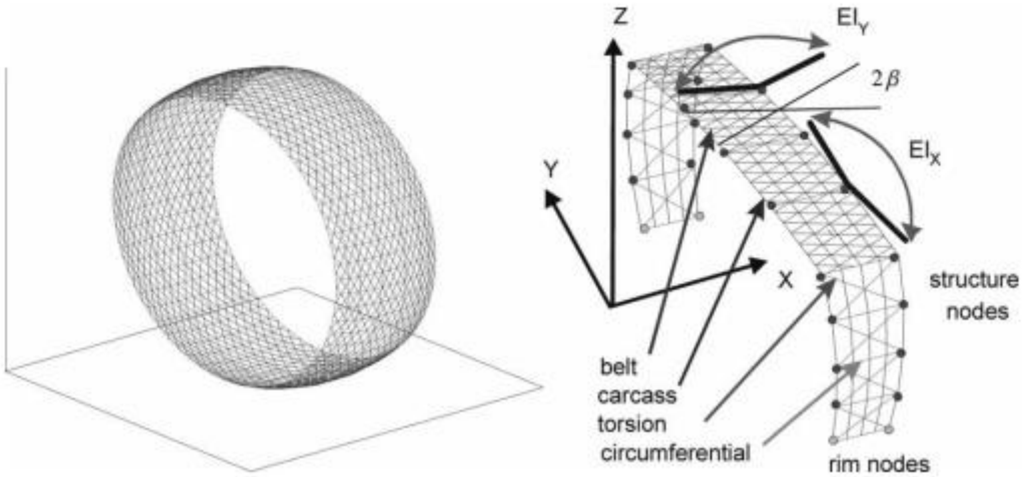

Figure 2 RMOD-K belt mesh and cross-section element (Lunger, et al., 2007)

A set of input parameters are required to calculate the response which includes General data (Dimension and mass of tyre), Topology Data (Cross section, belt angle), forces and moments Stiffness etc.

The main advantage of this tyre model is that it considers the surface's unevenness, while the requirement for higher computational power is a major drawback.

Tame tyre

This model provides a detailed description of a Tyre's thermal and mechanical attributes when calculating the longitudinal and lateral forces along the moments acting on the Tyre under pure and combined slip conditions at multiple loads, inflation pressures, camber angles, ambient temperature etc.

The Tame Tyre model is based on the physical characteristics of the tire that focus on different aspects such as the Tyre mechanics, rubber material properties and temperature parameters, containing important information such as the description of the contact patch, the carcass, the stiffness, and the thermal model throughout a rotational period.

The tread surface, belt, and internal tyre temperatures are derived using a one-dimensional thermal equation in thread thickness averaged over the tyre's width and the total rotation time.

In this model, assumptions are made considering characteristics such as the Tyre's uniformity, and its movement over flat surfaces. Additionally, the forces calculated are only for dry weather conditions and for a limited frequency range. The thermal characteristics like the heat flux are considered to be homogenously spread over the tyre surface for a single period of rotation.

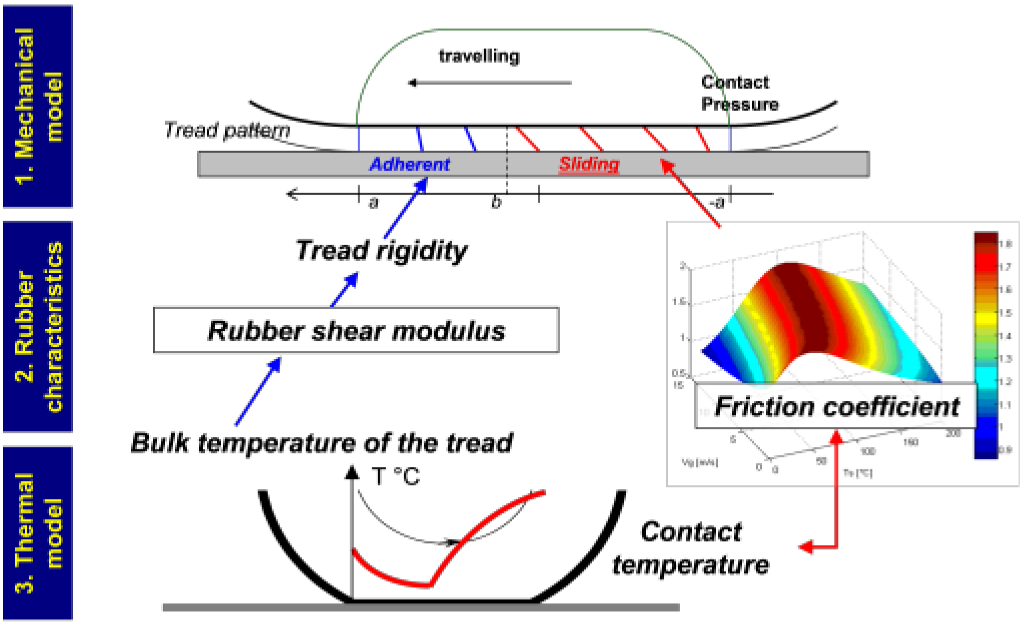

Figure 3 Tame Tyre's overall description (Michelin, n.d.)

The input parameters required are travel speed, inflation pressure, outside air temperature, track temperature, time, initial tyre temperature, slip ratio, side slip angle, camber angle, and load.

The available output is the longitudinal and lateral tyre forces, self-aligning torque, surface temperature, belt temperature, and internal temperature of the tread.

This model helps analyse and predict the Tyre and vehicle performance for any type of operating conditions which remains important means for the improvement of Tyre models for handling and vehicle dynamics that influence suspension design and chassis set-up.

This model enables fast calculations of tyre forces and moments to be performed for vehicle dynamic simulation with a capability for real-time simulation.

Comparison Sheet

Contents	F Tyre	RMOD-K	Tame Tyre
Focused on	Comfort and durability	Comfort and durability	Driving dynamics
Advantage	Can be used for wide-ranging applications and can be used in 3D cases. Simple tests can be carried out swiftly. Thermal function to inspect friction and pressure changes	A wide range of applications, can be used for different tyre models of different complexity. Considers irregularities of the road surface.	This model in comparison to the magic formula can deduce Tyre forces and moment of the driving conditions of the vehicle. The model is powerful to predict the Tyre and vehicle performance for any type of manoeuvres, circuit design and track properties.
Accuracy	Up to 120 Hz	Up to 100Hz	
Capability	Used for steady-state handling analysis, ride comfort analysis and road load test.	Used for cases ranging from steady state to complex 3D high-frequency short wavelength cases.	Used to predict tyre behaviour considering mechanical and thermal properties.
Application	These models can be used to predict the performance on different road condition temperatures and also determine the optimal pressure and camber of the tire for maximum performance Ideal for Improving performance, efficiency and safety	This model is used to study dynamic behaviour, the interaction between the tyre and the road surface, etc. It is also used to simulate the effects of driving scenarios and can also identify the parameters such as stiffness and damping of tires.	It is used in the design optimization of vehicle dynamics, tyre models and road interaction. This model can also be used for simulating tyre dynamics in racing cars and performance vehicles

Table 1 Comparison of Tyre Models

Section – 2 Suspension Optimisation

Introduction

Suspension optimisation is a very critical component in vehicle dynamics. It helps to improve the performance of the vehicle's suspension system and make it more responsive to the movement of the road. Good suspension optimisation is necessary to achieve good ride comfort and improve handling characteristics. Suspension optimisation can be performed by adjusting the spring rate, damping rate, ride height, camber angle etc.

In this suspension optimization exercise, we have modelled and simulated an SDOF, 2DOF and 4DOF lumped parameter model using Adams software and have discussed basic optimization principles to enhance suspension performance.

Suspension optimisation using SDOF

In the single DOF model, a larger approximation is made, and it is assumed that the four suspension wheels are integrated into one and function as one, with the damper showing only translation motion. The upright is not taken into account in this case, nor are the roll and pitch motions. The model consists of two blocks representing the vehicle body and the road, which is associated with a spring damper. The input from the road is given by using the road profile data.

Vehicle Mass	906kg
Spring Stiffness	80 N/mm
Damping rate	0.8 N-sec/mm

Table 2 Input parameters for the SDOF model

Optimisation of the suspension can be carried out in two different approaches, one by keeping the coefficient of damping constant and varying the spring stiffness, while for the second approach the spring stiffness is kept constant and the damping is varied. To measure the performance of the suspension system we have created two measure quantities to determine and compare the motion of the upper block in response to the input.

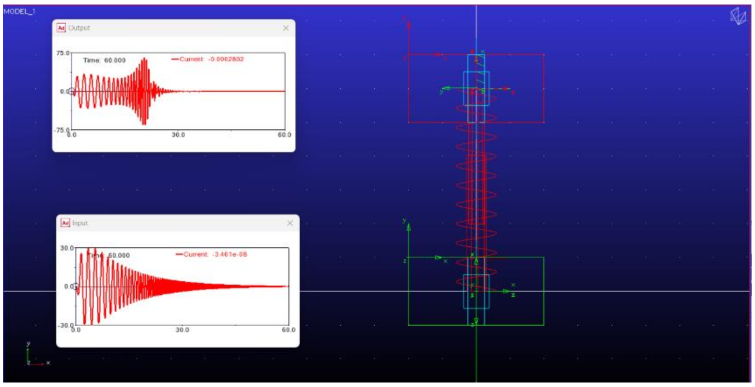

Figure 4 SDOF model response

From the output graph in the above image, we can see the behaviour pattern of the body in response to the input, a sudden rise in the amplitude in contrast to the input waveform indicates the resonance region after which the amplitude decreases and shows a direct response once again.

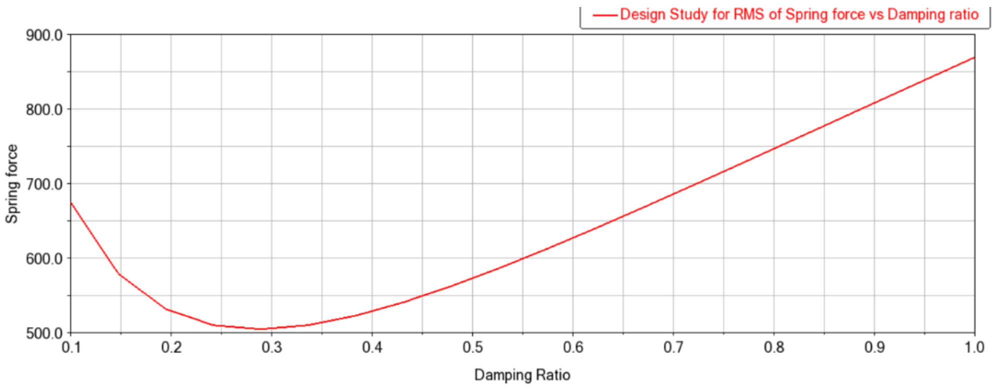

Figure 5 Graph of RMS of spring force vs damping ratio

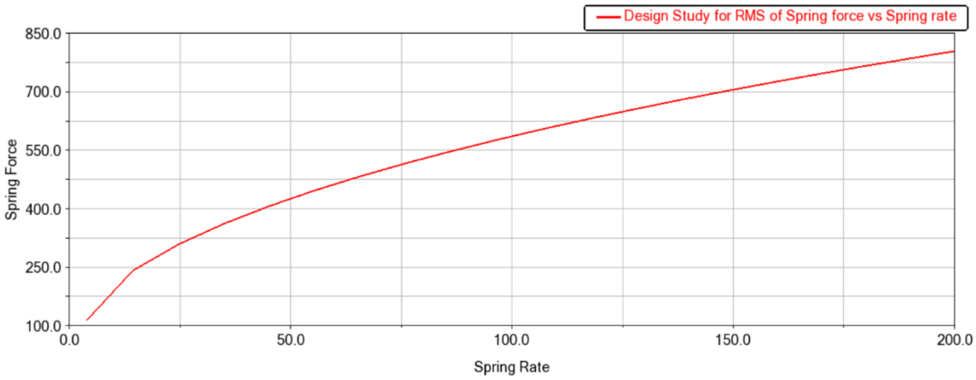

Figure 6 Graph of RMS of spring force vs spring stiffness

From the above graph, the design study for optimization of damping value suggested the ideal damping ratio of 0.29. Following the spring design optimisation study, we can say softer the spring better the suspension, and we can conclude that the lowest possible spring rate is required to minimise the contact patch variations.

Suspension optimisation using 2DOF

The 2DOF model is a slightly increased complexity model in comparison to the SDOF model. This model allows us to consider the system's upright, which gives us a more in-depth knowledge of the suspensions' behaviour. As the model allows for optimisation in response to body acceleration and contact patch load, it becomes useful in improving the ride and handling performance.

For the 2DOF system, the top block represents the vehicle mass, whereas the middle block acts as the upright of the car which wasn't considered in the SDOF system and the lower block models the road. The upper spring represents the suspension, and the lower spring represents the tyre, which has a lower damping value. For the simulation of this two-degree-of-freedom model, we included a road profile as the input waveform.

In the 2DOF we can get two separate modes of vibration.

Parameters	Value
Vehicle Mass	906 kg
Upper Spring Stiffness	80 N/mm
Upper Damping Co-efficient	0.8 N-sec/mm
Lower Mass	90 kg
Lower Spring Stiffness	150 N/mm
Lower Damping Co-efficient	0.12 N/mm

Table 3 Input Parameters for the 2DOF model

The bodes plot below shows the frequency response of the system. This suggests the displacement in response to the input. The goal of the optimization is to smooth the plot's curve. This intern will assist in determining the best value for the spring stiffness and damping coefficient.

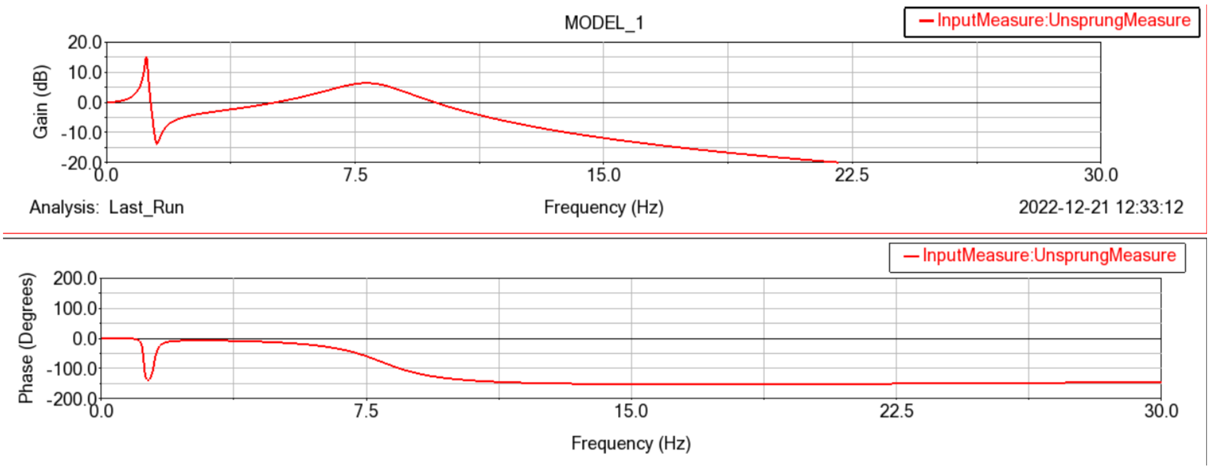

Figure 7 Bode plot for 2DOF system

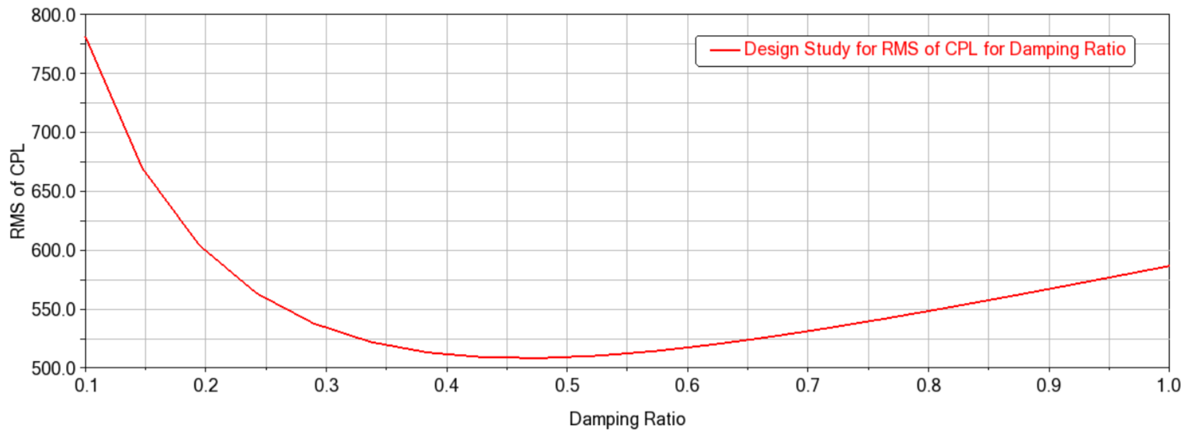

Figure 8 Graph of RMS of CPL vs damping ratio

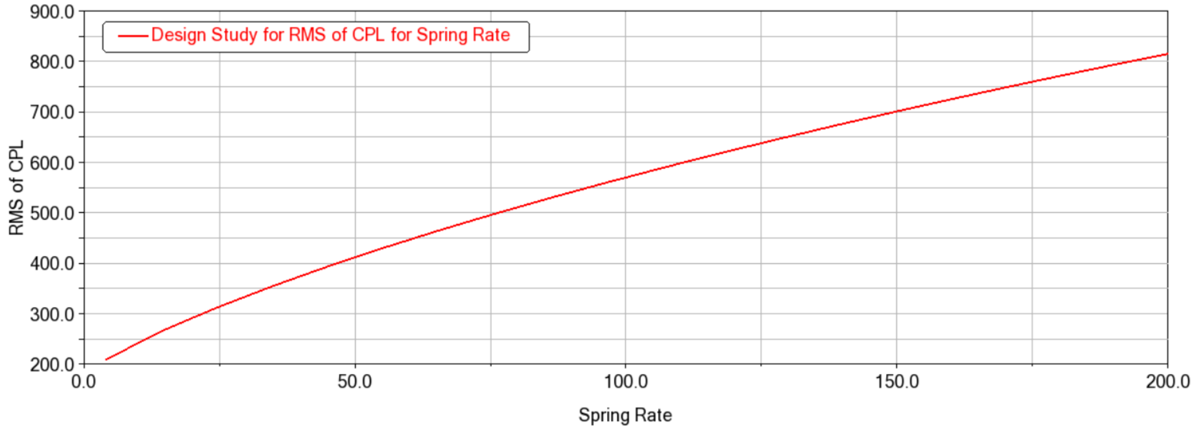

Figure 9 Graph of RMS of CPL vs Spring rate

From the above graph, it can be predicted that for the used road profile the ideal damping ratio is approximately 0.47 to obtain the minimum variation in the CPL which is important for better grip. Also with an increase in spring stiffness, the influence of the suspension reduces so it is ideal to select a lower value of the spring rate.

We can also compare the results of the optimisation study done for both contact patch load and body acceleration, based on the characteristics required of vehicle and ideal value of damping can be selected.

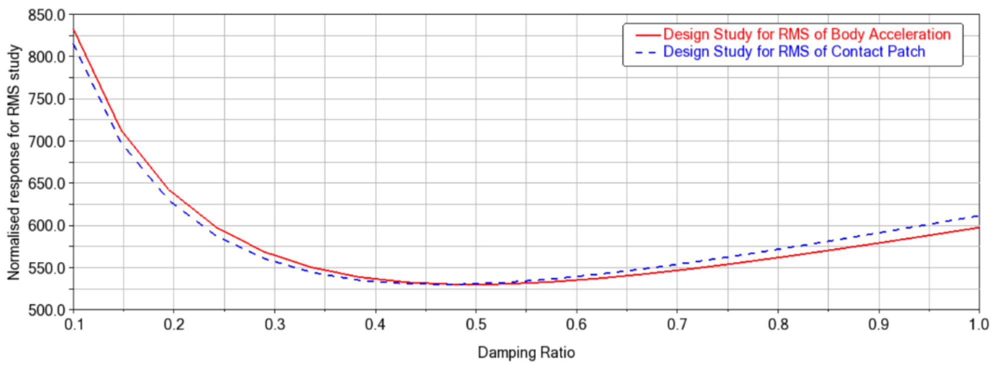

Figure 10 Comparison of Damping optimisation for Body acceleration and Contact patch load

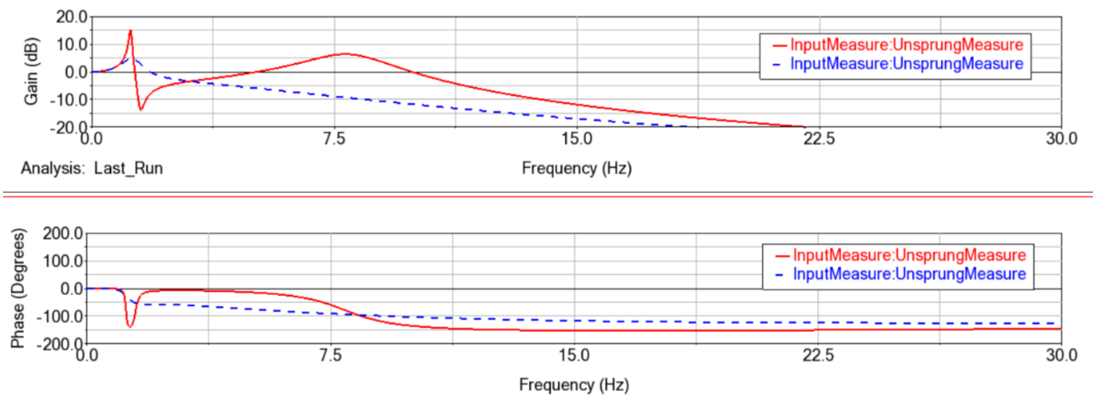

Figure 11 Bode plot comparison for initial and optimised values

The above bode plot suggests the reduction in the variation using the optimal damping.

Suspension optimisation using 4DOF

For a 4DOF system, we have extended the model by considering two wheels with upright. The vehicle body is connected to these uprights having both springs and dampers. In comparison to the 2DOF, this body is free to move vertically as well as rotate about its COG. This model is used to understand the yaw, roll and wrap conditions and the influence of suspension.

Using the 4DOF model, we can model roll in the front view and pitch in the side view. We have considered the side view of the vehicle and we have the front wheel and rear wheel combination. The model is used for the analysis for heave input. In this analysis the consideration of polar moment of inertia is essential, this was not possible with 2DOF model.

When working with the 4DOF model from a side view, for example, for the pitch model, we can specify a time delay in the inputs to make it realistic like the rear wheel going over the pothole with a time gap after the front wheel.

Mass	906kg
Wheelbase	2.6m
Distance of CoG from front axel	1.2m

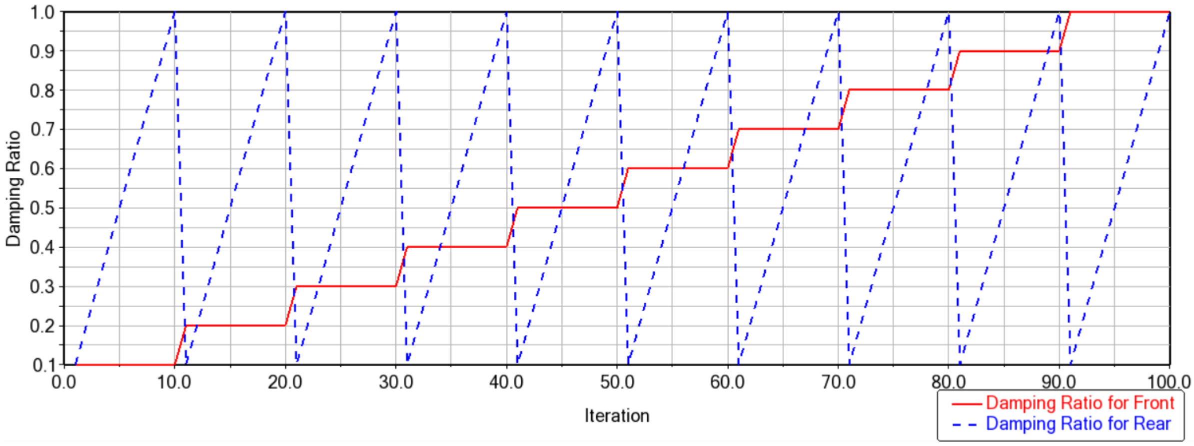

Figure 12 Damping ratio optimisation for different trails

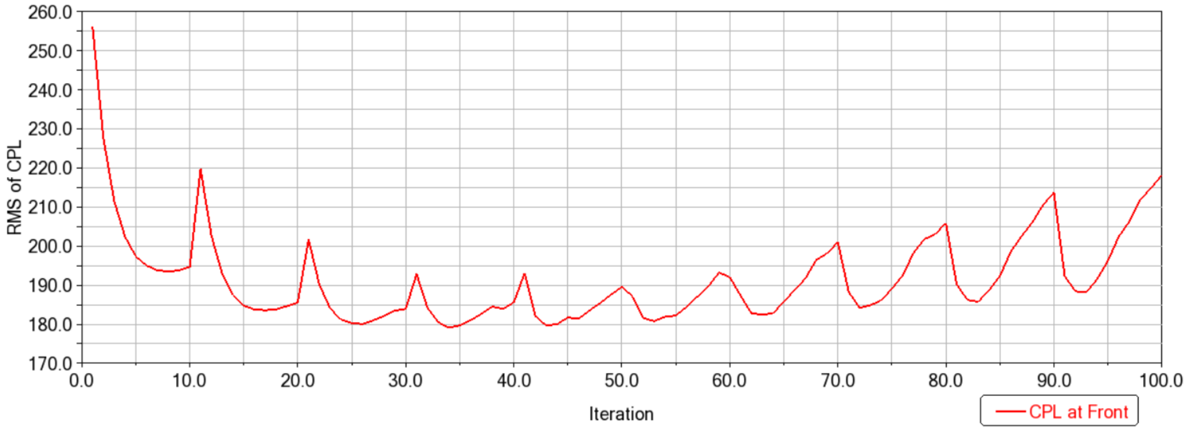

Figure 13 RMS of CPL for different trail

Above plots represents the optimisation of damping ratio for multiple iterations for both front and rear suspension, suggesting an optimised damping ratio of 0.5 and 0.4 respectively.

Conclusion

The optimisation of the suspension system is not accurate using the lumped parameter model for a lower degree of freedom model because of the simplicity leading to a larger approximation. With the increase in the degrees of freedom allows us to understand and study the effect of the suspension in a more detailed manner.

Since the following techniques are conducted for sinusoidal waveform as an input function for road profiles which are different from real-world scenarios so it is difficult to optimise the suspension for true conditions. It is difficult to create a road profile with different bump profiles.

Section - 3 Derivatives and Bicycle Model

Introduction

In the following exercise, we have created a spreadsheet using derivatives to calculate the yaw rate in response to a Step-Steer input using parameters of neutral steering and a critically damped vehicle model. This obtained response was compared with the normalised response achieved from the simulation of the bicycle model developed using the Adams software.

For a car to have neutral steering and be critically damped the following equation needed to be satisfied.

$$\frac{I}{m} = \frac{N_R}{Y_\beta}$$

The reference value used was,

Mass	900	Kg
Polar Moment of Inertia	1100	Kg.m^2
Wheelbase	2.6	m
Cornering Stiffness f	1100	N/deg.
Cornering stiffness r	900	N/deg.

Table 4 Reference Parameters for Derivative Analysis

Firstly, we achieved a neutral steering model using values a, b, C_f & C_r.

As,

$Under\ Steer\ gradient = \frac{m}{l} * (\frac{a}{C_r} - \frac{b}{C_f})$

For the under-steer equation to be zero,

$b * C_r = a * C_f$

Further to achieve the value of $\zeta = 1$, which is dependent on values a, b, C_f, C_r, mass of the vehicle and the moment of Inertia, either of mass or moment of Inertia could be adjusted. Since the Polar moment has a direct relation to multiple parameters the optimal strategy was to vary the mass using the above relation to achieve the required condition.

It was important to consider that the value of N_r (i.e. yaw damping) used to calculate the value zeta is velocity dependent.

Derivative Analysis

The derivative analysis approach allows us to solve the equation of motion to determine the relationship between the provided steering input and the yaw response of the vehicle.

Input Parameters

Mass	658	Kg
Polar Moment of Inertia	1100	Kgm^2
Wheelbase	2.6	m
Cornering Stiffness front, Cf	1100	(N/rad)
Cornering Stiffness rear, Cr	900	(N/rad)

Table 5 Input Parameters of the model

The above-listed parameters are used for both the analytical and numerical models (Adams model).

Response	Wheelbase (l)	COG from front axle (a)	COG from rear axle (b)	Damping Ratio
Underdamped	2.6	0.9	1.7	0.69
Critically damped	2.6	1.17	1.43	1.00
Overdamped	2.6	1.3	1.3	1.53

Table 6 Input Parameters for steering response

In the derivative analysis approach, six derivative terms are being used which have an important role in the dynamics of the vehicle. These six terms relate to the curvature, lateral acceleration, roll angle, slip angle, yaw rate and steering angle of the vehicle among which the term that signifies the lateral acceleration and yaw rate are velocity dependent and have an influence on the behaviour and response of vehicle with changing velocity.

Derived Parameters

Using the spreadsheet we have derived the yaw response with respect to time for all three steering responses.

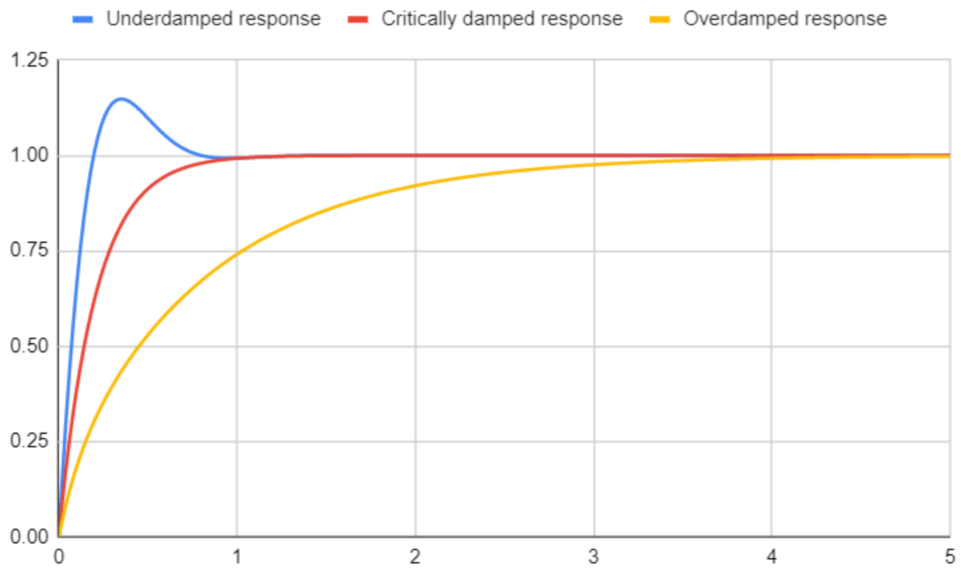

Figure 14 Normalised response using derivatives

From the above graph, we can observe an overshoot in the underdamped condition before settling which is because of the low damping ratio, whereas the overdamped takes much more time to reach the steady state. In contrast, the critically damped response exhibits a smooth response and requires a shorter duration to settle making it the most desirable response.

In the derivatives, only linear values of cornering stiffness are being considered, due to which the lateral force produced increases linearly with slip angle, and there is no limit behaviour, which is not the case in real-world scenarios.

ADAMS Model

As the second part of the exercise, a numerical model has been created in ADAMS using the same vehicle parameters as used for the analytical calculation in the spreadsheet. The bicycle model is made up of three connected parts: the front wheel, the chassis, and the rear wheel. The front wheel was attached to the chassis with a hinge joint to support steering action, whereas the rear wheel and chassis had a fixed joint. The main body and the ground had a planar joint defined.

Both front and rear wheels were specified with negligible mass and moment of inertia.

Forces & Motions

To specify the steering input, the vehicle has to maintain a constant speed. The model was described with two forces acting in the longitudinal direction to control the vehicle's motion. The specified forward force was 13000 N**,** and a state variable was defined for the drag force **(1e-4**VARVAL(Velocity))**2,** the variables were handy to attain a constant velocity of 80mph.

For the steering action, the front wheel was provided with a haversine step input function of 5 degrees for 0.01 seconds once the vehicle attained the required speed.

Haversine Step (time, 3.0,0.0d, 3.01,5d)

Further state variables were created to determine the Slip angles for the front and rear wheels and were used for the defining lateral force function.

Result

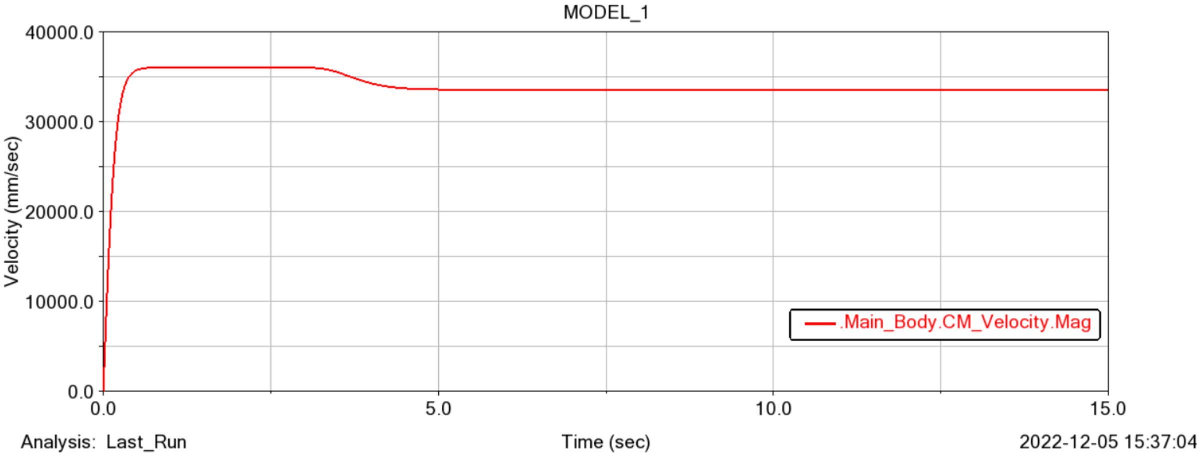

Figure 15 Speed vs time graph suggesting influence of steering input

The above graph suggests a decrease in the vehicle speed when the steering input is provided. This speed reduction occurs due to the component of lateral force which is acting in opposite direction of travel.

Comparison between Adams model and Derivative Analysis

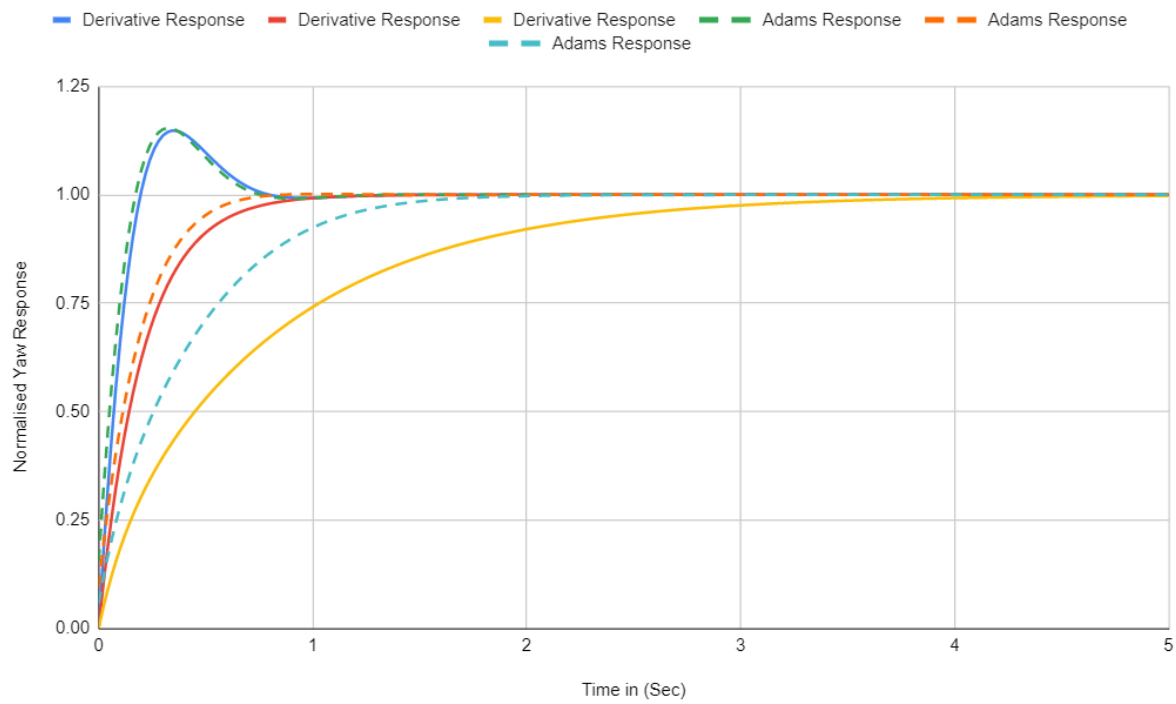

Figure 16 Comparison graph between derivative and numerical model response

The above graph shows the comparison of the normalised response obtained from both approaches, we can observe a similar trend but with variations in the results

Possible reasons for variations in the two results

- In the derivative approach the results obtained are for constant velocity which is not the case in the Adams model where the speed decreases with steering input due to the consideration of real-time force.
- The inertia value being considered in Adams's model
- The steering input provided in either approach.
- The errors involved in considering the decimal values in the derivatives

By using non-linear tyres in the Adams model we can achieve the limit behaviour for the vehicle as the force on the tire will automatically reduce once the maximum lateral force is attained.

Conclusion

The Lateral acceleration, slide slip angle, yaw rate, steering angle, and tyre slip angle are velocity-dependent parameters that affect a vehicle's cornering performance and vary depending on the vehicle's velocity. They are important in cornering analysis as they determine the vehicle's behaviour to the forces acting when the cornering action is performed.

The analytical method provides a good approximation of the vehicle behaviour and is handy for vehicle dynamists at an early stage of designing, but for accurate results, it is preferred to use the numerical approach which can be validated using the derivatives.

References

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